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EXPERIMENTAL WINDAGE LOSSES FOR A LUNDELL GENERATOR OPERATING IN AIR IN THE TURBULENT-FLOW REGIME

by Sol H. Gorland and Erwin E. Kempke, Jr.
Lewis Research Center
Cleveland, Ohio
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ABSTRACT

Viscous torque (windage) of a Lundell-type rotor having a major diameter of 8 inches and rotating within a concentric housing was measured at speeds up to 36,000 rpm. Radial air gaps of 0.039 and 0.080 inch, corresponding to gap-to-radius ratios of 0.01 and 0.02, were tested. The 8-inch diameter cylindrical section of the housing was tested with a smooth inner surface and with an inner surface having axial slots to simulate the winding slots in an actual alternator. In the turbulent-flow regime, power loss for a slotted housing is 30 to 40 percent higher than for a smooth housing. Pressure rise due to the pumping effect of the conical sections was measured; this pressure rise can be calculated using equations for enclosed rotating discs.

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SUMMARY

Viscous torque (windage) of a Lundell-type rotor having a major diameter of 8 inches and rotating within a concentric housing was measured at speeds up to 36,000 rpm. Radial air gaps of 0.039 and 0.080 inch, corresponding to gap-to-radius ratios of 0.01 and 0.02, were tested. The 8-inch diameter cylindrical section of the housing was tested with a smooth surface and a surface machined with axial slots to simulate the winding slots in an actual alternator. Power loss for a slotted stationary cylinder operating in the turbulent flow regime is 30 to 40 percent higher than for a smooth cylinder. Pressure rise due to the pumping effect of the conical sections was measured. It was found that pressure rise for a rotating conical section can be calculated using equations for enclosed rotating discs.

INTRODUCTION

The Lundell generator is being investigated for use in Brayton-cycle space power systems (ref. 1). At the speeds involved (24,000 rpm and 36,000 rpm) windage loss becomes high and results in significant heat flow in the electrical generator. This effect must be included in the generator thermal design.

Turbulent velocity profiles developed in the rotor-stator gap represented by Reynolds numbers up to 100,000 are well above the levels previously investigated. Taylor, Vohr, and Wendt (refs. 2, 3 and 4) have studied turbulent flow (Reynolds numbers above 7000) for concentric rotating cylinders with Reynolds numbers as high as 40,000. These high Reynolds numbers, however, were obtained using large gap widths; e.g., gap-to-radius ratios of 0.1 to 0.23 (refs. 2 and 3). Reference 5 presents a preliminary study of windage losses for concentric rotating cylinders with Reynolds numbers as high as 100,000 using gap-to-radius ratios of 0.01 to 0.04. These are the gap sizes considered for designs of space-power alternators.

Studies of enclosed discs rotating in the turbulent flow regime can be found in references 6 and 7. The gap-to-radius ratios for these investigations ranged from 0.01 to 0.22. No information was available on enclosed rotating conical sections. Experimental data were needed to extend the existing information for incorporation into the design of future electrical generators.

A Lundell-type rotor having a major diameter of 8 inches was rotated at speeds from 0 to 36,000 rpm. Two gap widths, 0.039 and 0.080 inch (gap-to-radius ratios of 0.01 and 0.02) were tested. The housing (stator) enclosure was mounted on a reaction-torque device so that viscous drag from the rotor would be measured. The housing was made in five separate and rigidly connected sections corresponding to the change in cross section of the rotor. The 8-inch diameter cylindrical section of the housing was tested with a smooth surface and a surface machined with axial slots to simulate the winding slots in an actual alternator. All tests were run in ambient air.

APPARATUS AND PROCEDURE

Rotor-Housing Configuration

The test unit (fig. 1) consisted of a rotor mounted on air-oil-mist lubricated ball bearings and a housing (stator) attached to a "floating" support table. Strain gages mounted in four flexure arms held the housing and support table so that it would pivot about the rotor axis. Reaction torque (viscous drag) between the rotor and housing was measured by means of a Wheatstone bridge circuit connecting the strain gages. A variable-speed dynamometer consisting of a dc motor and two tandem gear units was used to drive the rotor. A splined coupling connected the rotor to the drive system.

Dimensions for the Lundell rotor shown in figure 1 are given in figure 2. The rotor was made from a heat-treated forging of a low-alloy vanadium steel. Surfaces were ground to less than a 16-micro-inch surface finish.

Testing Sequence

The aluminum housing consisted of five sections, each split axially so that it could be mounted or removed without affecting the test rotor (fig. 1). Tests were performed on the center cylindrical section and the complete housing assembly. All parts were doweled or keyed together so that the machined alignment of the parts could be reproduced. After a complete series of tests had been performed on a set of smooth-surface housings, the center cylindrical section of the housing was remachined with axial slots to simulate alternator-winding slots. These slots were 0.119 inch wide by 0.010 inch deep spaced 6 degrees apart for a total of 60 slots. The entire housing was then remachined to a larger radial gap for the second series of tests with a smooth housing. Then axial slots were again machined into the main cylindrical section for second series of tests with a slotted housing. The two radial gap sizes used were 0.039 and 0.080 inch.

The rotor was run at speeds up to 27,000 rpm for the smaller gap and 36,000 rpm for the larger gap. Excessive vibration amplitude measured early in the test program limited the peak speed.

Data measured several times at the same test condition were compared to verify reproducibility of the data. A test consisted of measuring torque, speed and temperature for the portion(s) of the housing mounted on the reaction torque device. For the 0.080 inch gap, pressure measurements were also taken to determine the pumping effect (pressure rise) of the conical sections. Differential pressure was measured with both conical sections mounted, deadhead pressure rise. All tests were performed in ambient air.

INSTRUMENTATION

Instrumentation consisted of rotational speed, torque, temperature, and differential pressure sensors.

Speed was measured by means of a magnetic pickup and a 60-tooth gear on the shaft of the dynamometer. The signal generated was sent to a counter and recorded. Rotational speed could be controlled within 5 rpm.

Torque measurements were made using a reaction torque device (fig. 1). Strain gages are located in four flexure arms so that they sense torque about only one axis. This axis was made to coincide with the axis of the rotor and its housing. Any torque developed about this sensing axis will produce a strain proportional to the torque. All tare torques or loads remain constant and are compensated for by calibration. The torque unit was calibrated by hanging accurately known weights from a calibration arm to produce known torques. Measurements were repeatable within 0.03 in-lb, although measurements could be taken at less than 0.01 in-lb increments. The strain gage output from the Wheatstone bridge circuit was measured on an integrating digital voltmeter. A 1 to 2 percent variation existed in the signal output. Torque measurements taken at 8000 rpm are approximately 5 percent accurate. The accuracy increases with the speed.

All temperatures were measured using iron-constantan type J thermocouples. A total of 13 thermocouples was mounted on the housing, each extending halfway into the gap. The thermocouple holes in the housing inside diameter were 0.040 inch.

A 2.5-psi differential pressure transducer was used to measure the pumping effect of the conical sections. The location of the pressure taps is shown in figure 2. The pressure transducer was calibrated against a precision Bourdon tube pressure gage. An integrating digital voltmeter measured the signal emitted by the transducer.

DISCUSSION OF RESULTS

The test section of the apparatus consisted of a smooth Lundell-type rotor rotating within an open-ended stationary concentric housing.

Viscous drag force, F, may be expressed as follows:

$$F = \lambda (A) \quad \left(\frac{\rho U^2}{2} \right) \qquad (1)$$

where
λ = drag coefficient
ρ = density
U = surface speed of rotor
A = surface area of rotor

The effect of pressure drag force is assumed negligible (ref. 8). For a cylindrical rotor, this expression may be rewritten as

$$\lambda = \frac{T}{\rho \pi \omega^2 R^4 L} \qquad (2)$$

where
T = torque
W = angular speed of rotor
R = radius of rotor
L = length of rotor

Nondimensional curves are plotted in figure 3 of drag coefficient vs. Reynolds number for the 8-inch cylindrical section. The curves presented are for the two gap sizes, 0.039 and 0.080 inch, with the slotted and unslotted housing. Here Reynolds number is defined as

$$Re = \frac{UD}{\nu} \qquad (3)$$

where the characteristic dimension D is the radial gap between stator and rotor, and ν is kinematic viscosity.

An assumption was made that the pressure in the gap remained ambient since the housing was open-ended. The temperature used to calculate Reynolds number was taken as the gap temperature in the center of the housing. The effect of using this hot-spot temperature, rather than an average, had a negligible effect on the curves. Thermocouple instrumentation indicated that the temperature profile for the axial length of the housing was symmetric.

Eight-inch Cylindrical Section

The plotted points for drag coefficient of the slotted vs. unslotted housing in figure 3 coincide for the lower Reynolds numbers. The curves for the 0.039-inch gap separate at an approximate Reynolds number of 2000 while the 0.080-inch gap separation point is approximately 4000. Both points correspond to approximately 3000 rpm. The increased power loss for the axial slots was approximately 35 percent at the higher Reynolds numbers.

Figure 3 shows that the curves of drag coefficient are within 5 percent of each other at the higher Reynolds numbers. This result is supported by Taylor in reference 2. Drag coefficients for the smaller gap sizes are lower than for the larger gap. A slope change for the smooth housing occurs at an approximate Reynolds number of 5000 to 7000. This is caused by the change from vortex to turbulent flow (ref. 3). The value of the slope changes from approximately -0.5, which correlates with references 3 and 4 for laminar flow to approximately -0.25 in the turbulent range.

Lundell-Shaped Housing

The plots of speed vs. windage for the complete rotor-unslotted housing combination are plotted in figure 4. Windage power loss, W, was calculated using equation (1). The plot for the 0.039-inch gap shows that windage is proportional to speed to approximately the 2.74 power while for the 0.080-inch gap the power proportionality is approximately 2.82. Windage is seen to be an inverse function of gap size.

Pump Characteristics of Conical Section

Pressure rise due to the pumping effect of the conical rotor sections was measured for the 0.080-inch gap housing. Differential pressure between the 5-inch diameter cylindrical section and the 8-inch cylindrical section was measured. Figure 5 presents the pressure rise (head) vs. speed curve and shows that the head rise varies as the speed squared. The deadheaded pressure rise can be accurately predicted using the method described in reference 7 for enclosed discs. The assumption is made that the average core velocity is one-half the rotor speed.

SUMMARY OF RESULTS

A Lundell-type rotor having a major diameter of 8 inches was rotated in air at speeds up to 36,000 rpm inside a stationary concentric housing. Two rotor-stator gaps were tested, 0.039 and 0.080 inches. The center cylindrical section of the housing was tested with a smooth surface and with axial slots to simulate alternator winding slots. Results of the test were:

1. Power loss for a slotted stationary cylinder operating in the turbulent flow regime is approximately 35 percent higher than for a smooth cylinder. Drag coefficients for the slotted cylinder coincided with the curves for the unslotted cylinder until approximately 3000 rpm.

2. The curves of drag coefficient vs. Reynolds number changed slope at approximately 5000 to 7000 Reynolds number, corresponding to the change from vortex to turbulent flow.

3. Power losses decreased as the gap size increased.

4. Pressure rise for a rotating conical section can be calculated using equations for enclosed rotating discs. The conical sections act as typical pumps, with the head rise proportional to speed squared.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 25, 1970.

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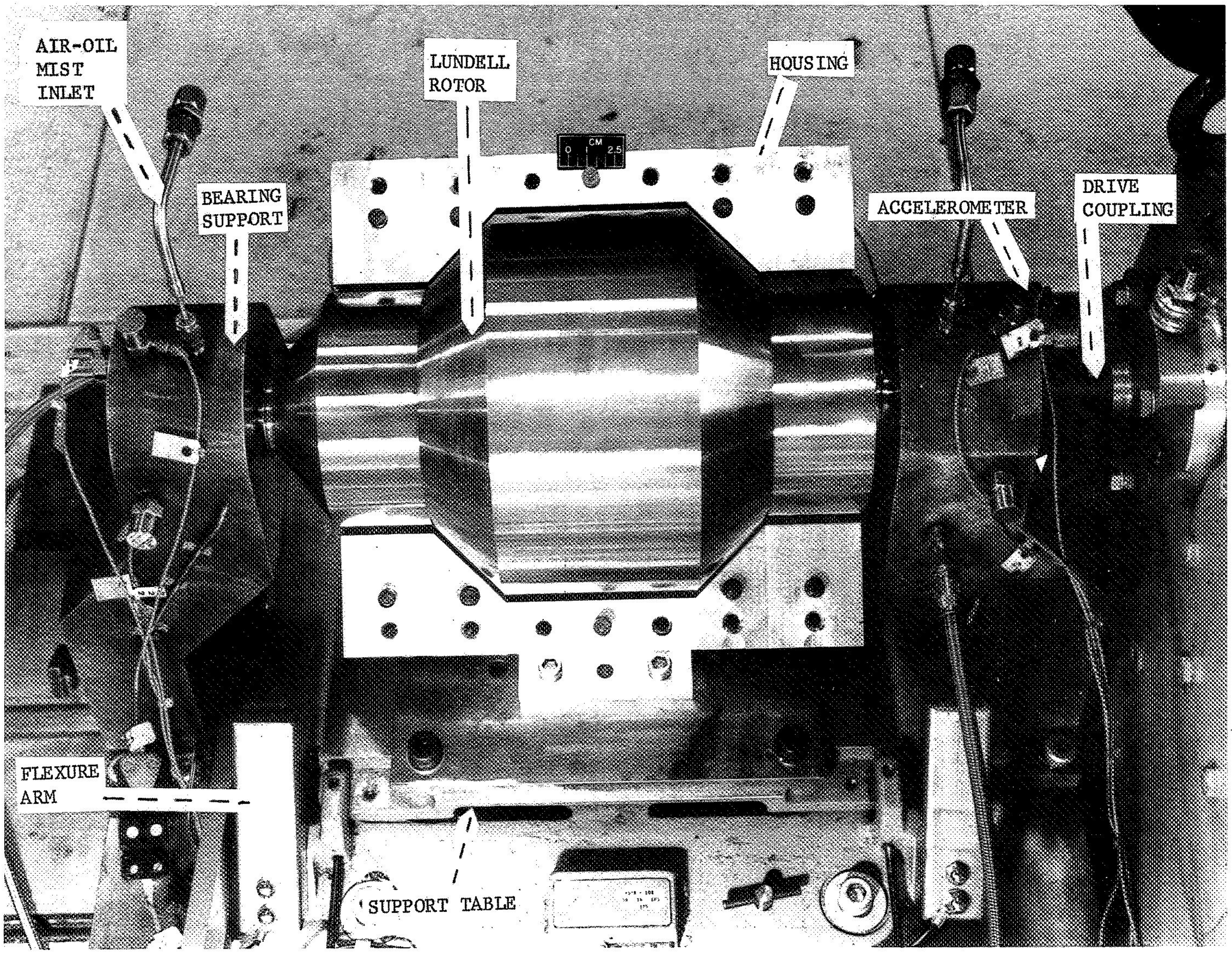

FIGURE 1. WINDAGE TEST APPARATUS FOR LUNDELL-TYPE ROTOR.

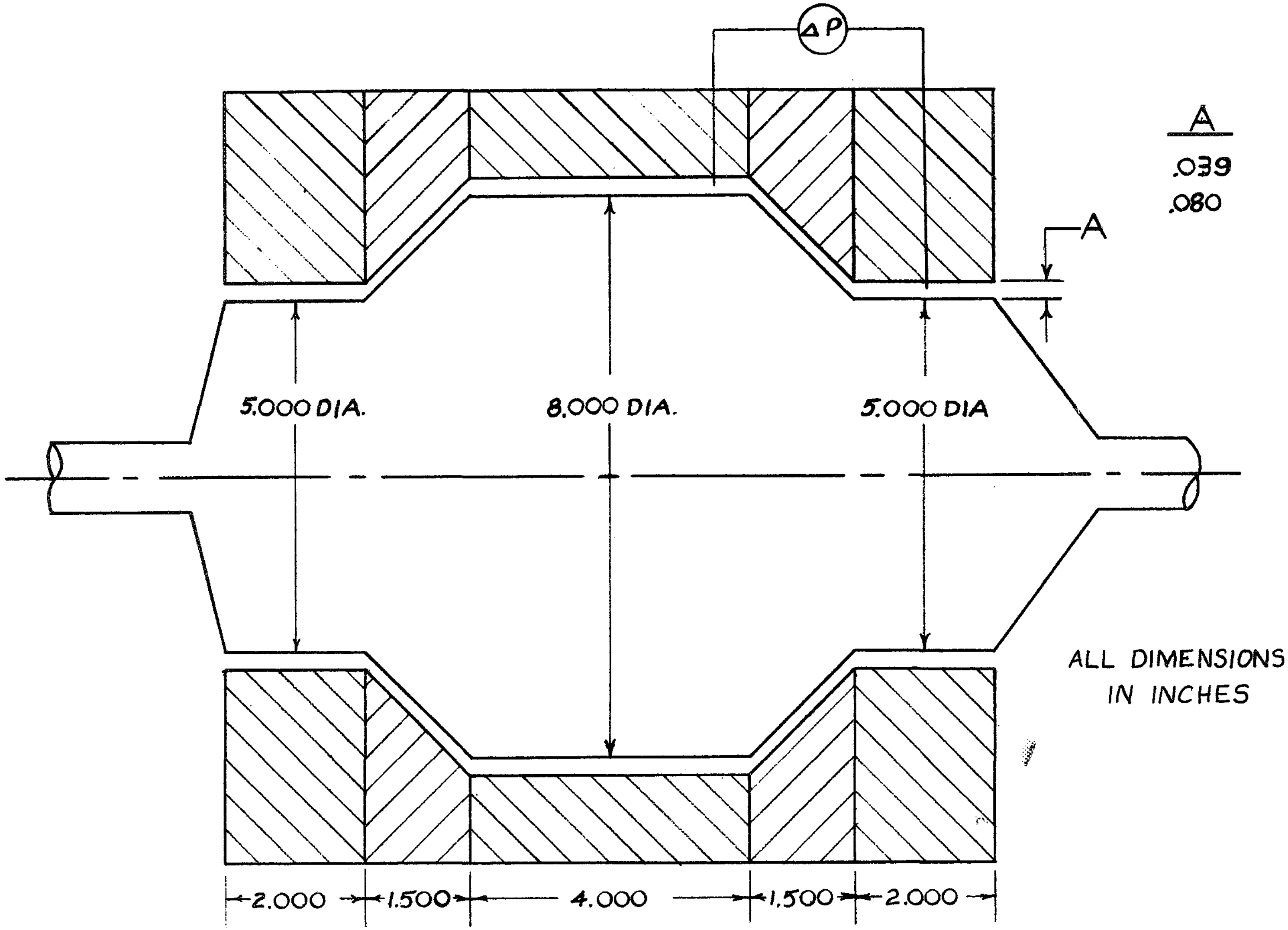

FIG. 2 ROTOR-HOUSING CONFIGURATION

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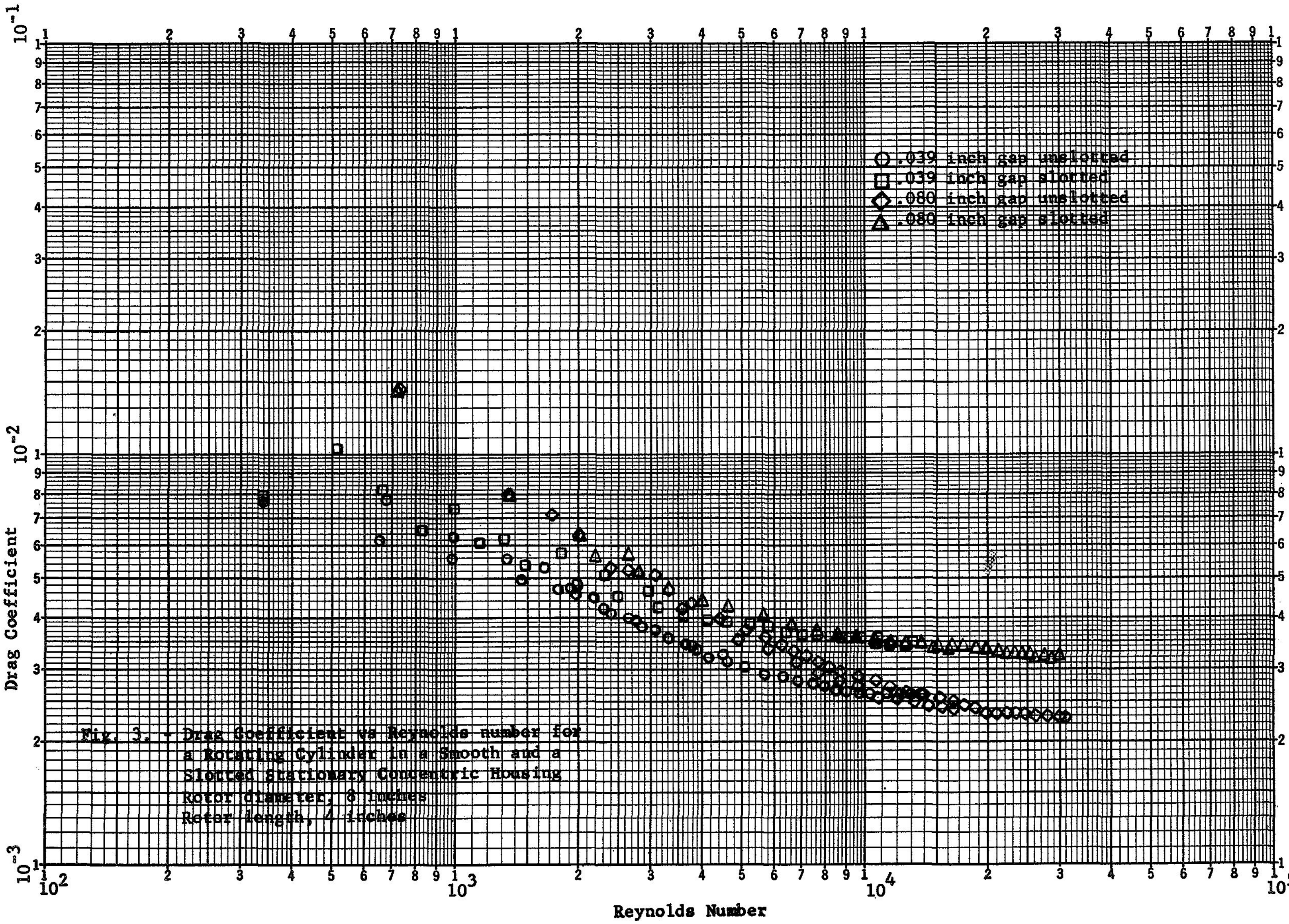

Fig. 3. - Drag Coefficient vs Reynolds number for a Rotating Cylinder in a Smooth and a Slotted Stationary Concentric Housing
Rotor diameter, 8 inches
Rotor length, 4 inches

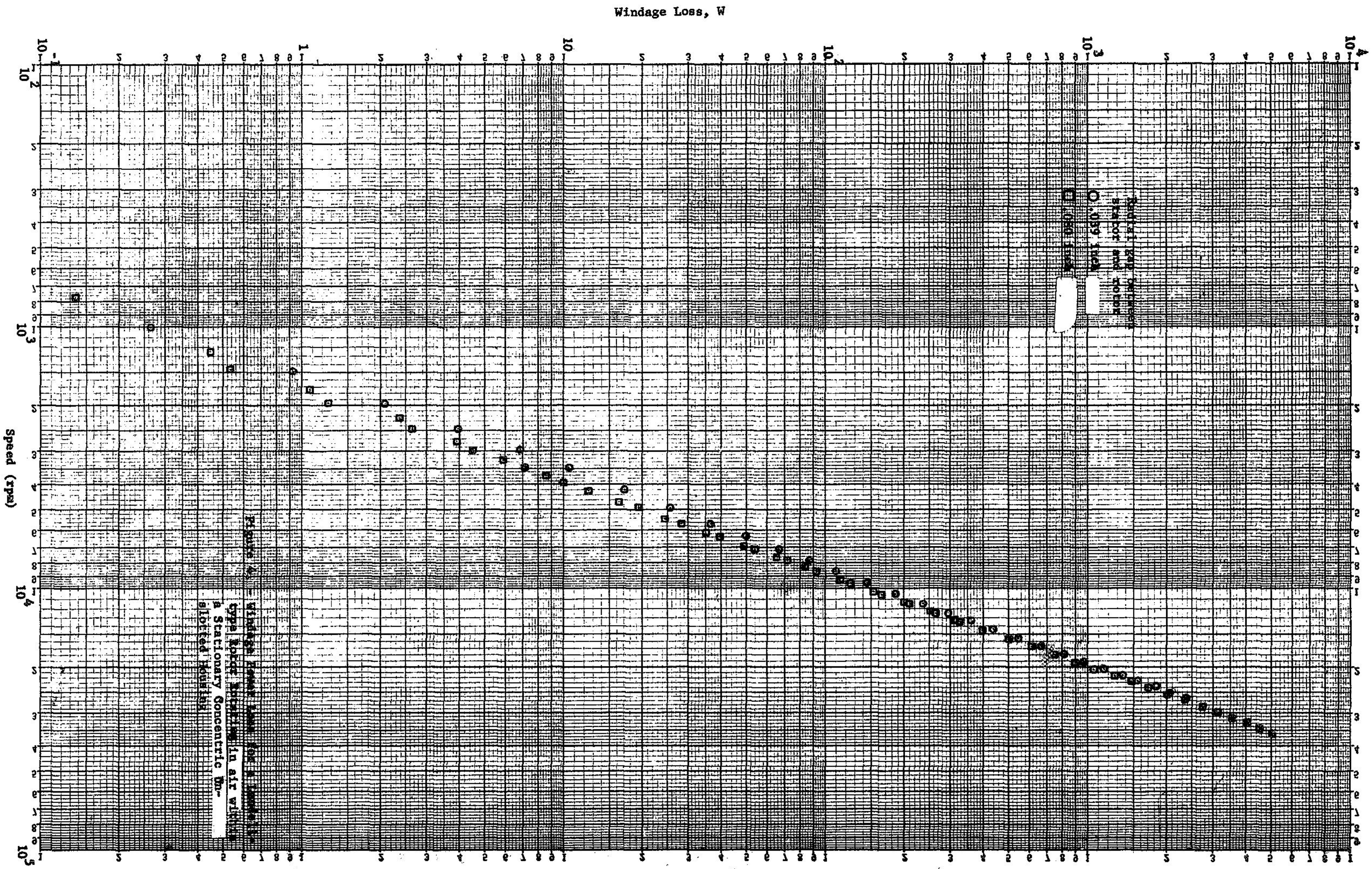

Figure 4. - Windage Power Loss for a Lundell-type Rotor Rotating in air within a Stationary Concentric Unslotted Housing

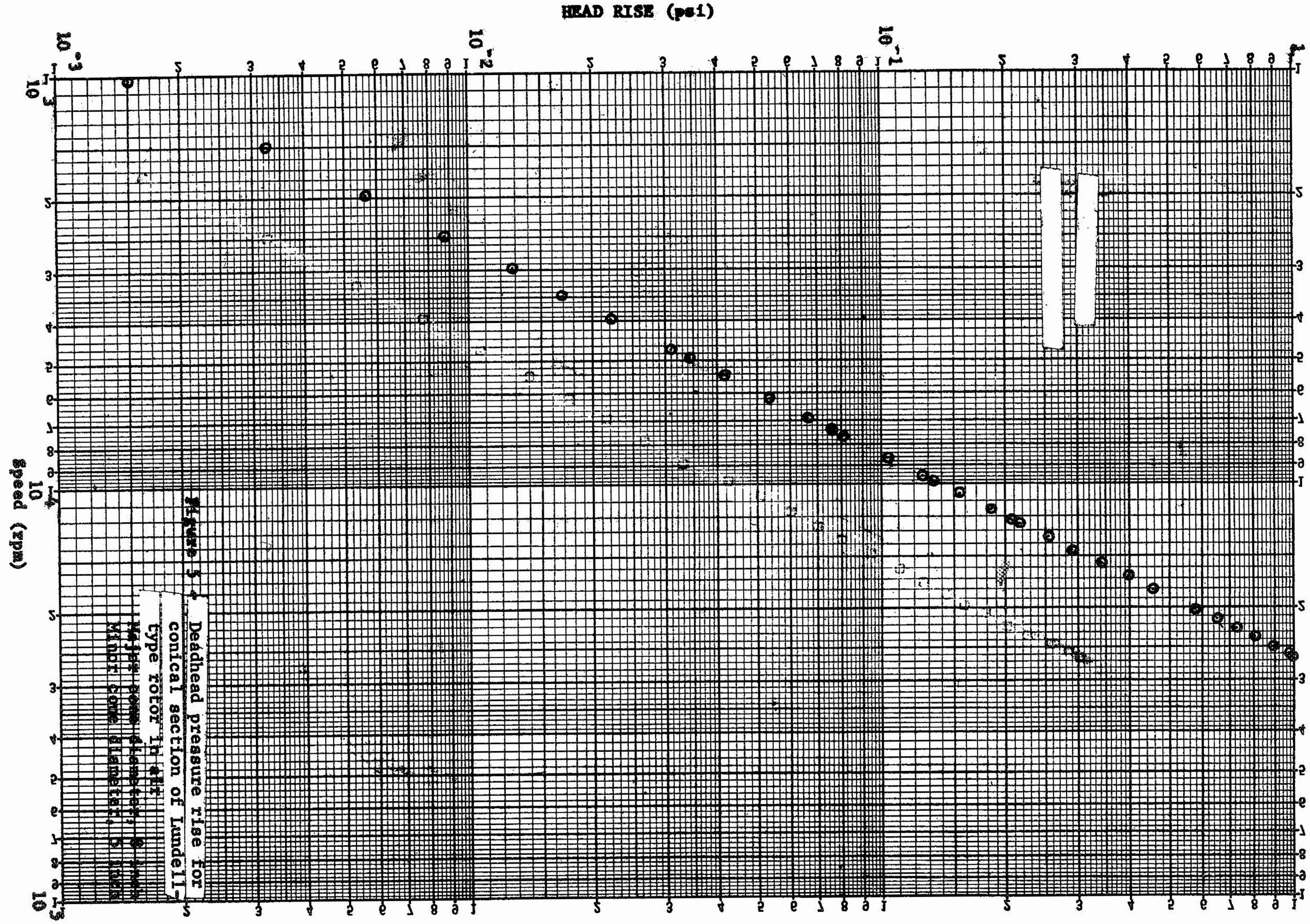

Figure 5 - Deadhead pressure rise for conical section of Lundell-type rotor in air. Major cone diameter, 8 inch; Minor cone diameter, 5 inch